(12) UK Patent Application (19) GB (11) 2 187 273 (13) A

(43) Application published 3 Sep 1987

(21) Application No 8526889

(22) Date of filing 31 Oct 1985

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(51) INT CL[4]
F23R 3/00

(52) Domestic classification (Edition I):
F4T 101 1112 1117 AK1
U1S 1987 F4T

(56) Documents cited
GB 1551860 GB 1383627 GB 0724721
GB 1524259 GB 1140757 GB 0699267

(58) Field of search
F4T
Selected US specifications from IPC sub-class F23R

(54) A gas turbine binary cycle

(57) In a gas turbine cycle, steam is injected into the primary combustion zone so as to surround the flame and exclude air from the part of the zone about the flame. The steam may be generated using the heat of the turbine exhaust gases. The combustion air is at least 3% above the stoichiometric amount.

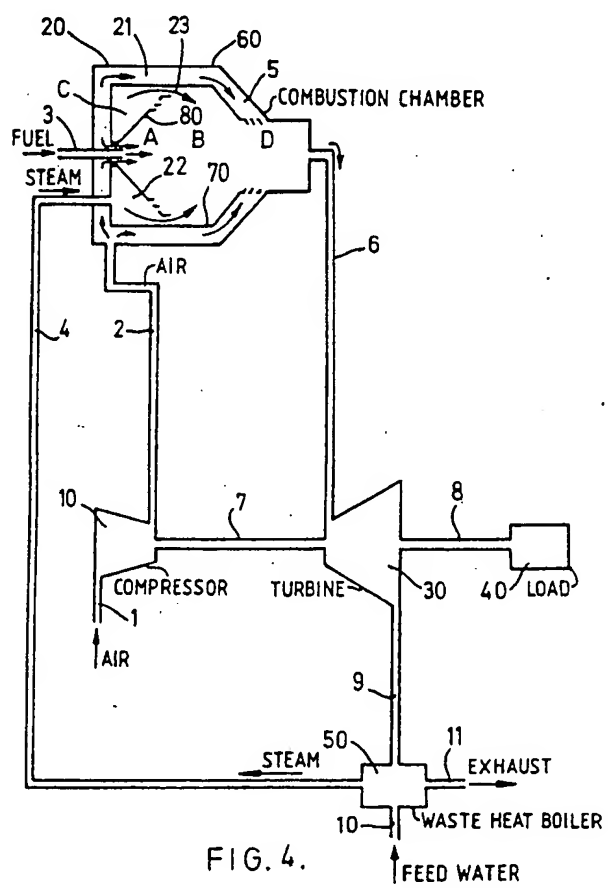

FIG. 4.

GB 2 187 273 A

The drawing(s) originally filed was/were informal and the print here reproduced is taken from a later filed formal copy.

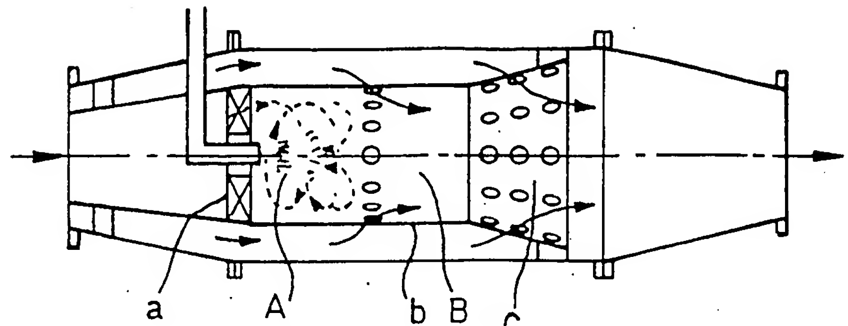

FIG.1.

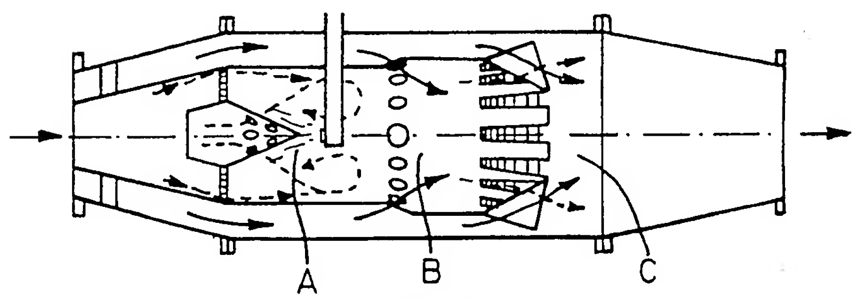

FIG. 2.

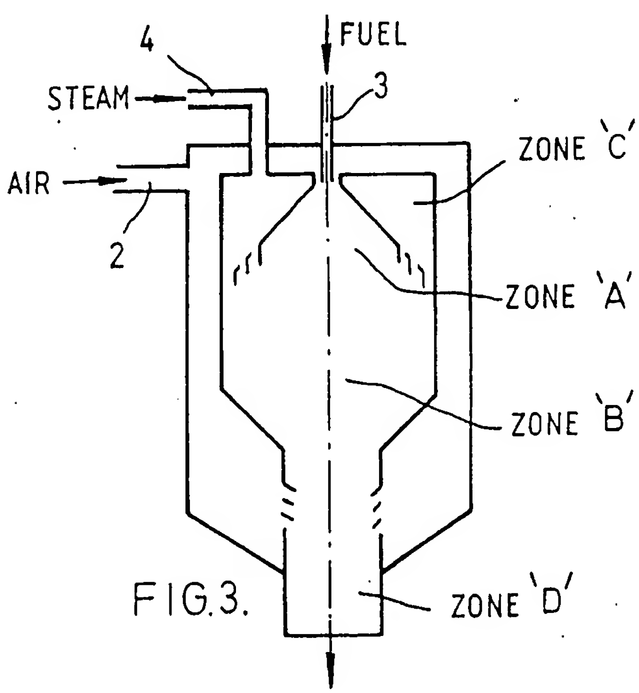

FIG.3.

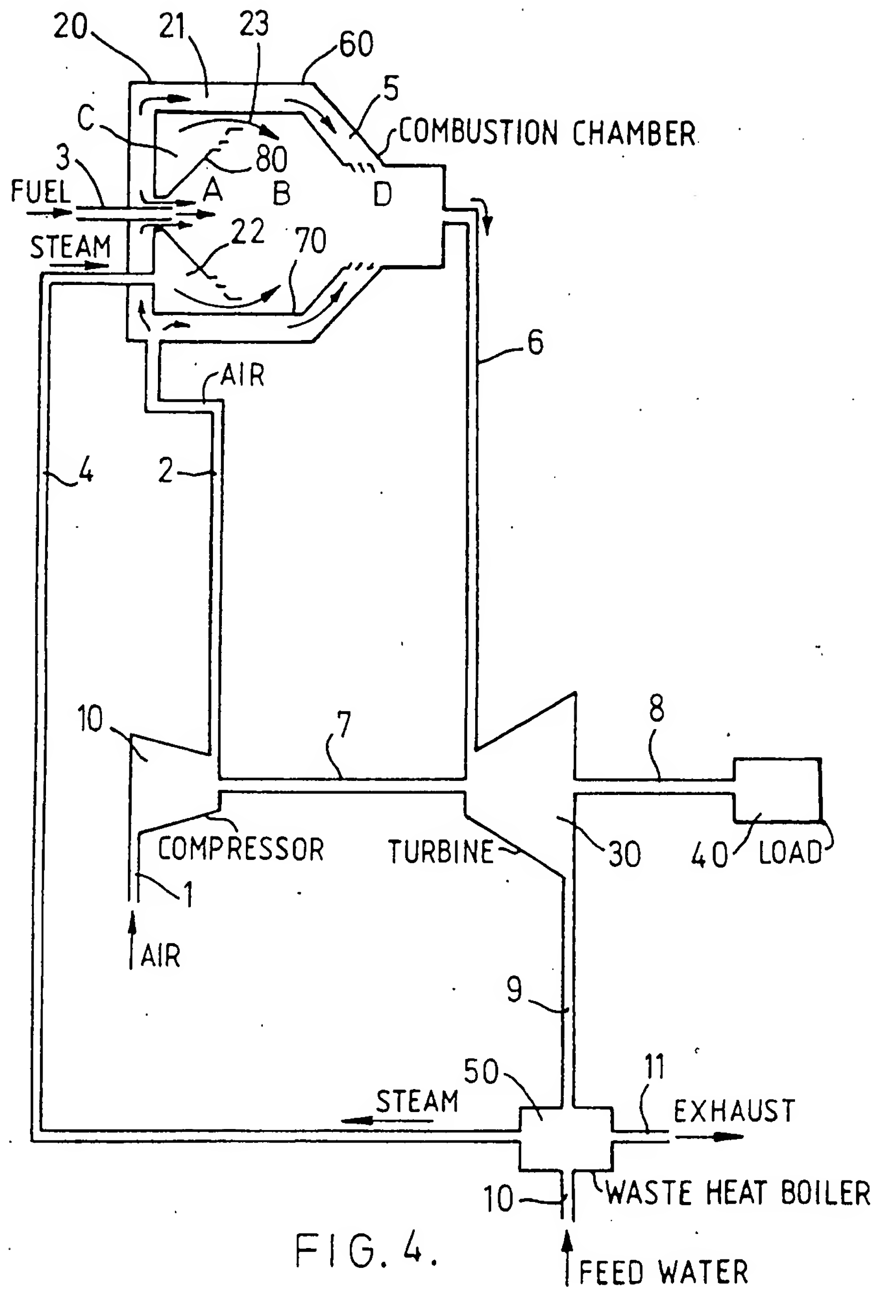
20
21
23
60
5
COMBUSTION CHAMBER
C
80
3
FUEL
A
B
D
STEAM
22
70
6
AIR
4
2
10
7
8
COMPRESSOR
TURBINE
30
40
LOAD
1
AIR
9
50
11
STEAM
EXHAUST
10
WASTE HEAT BOILER
FEED WATER

FIG. 4.

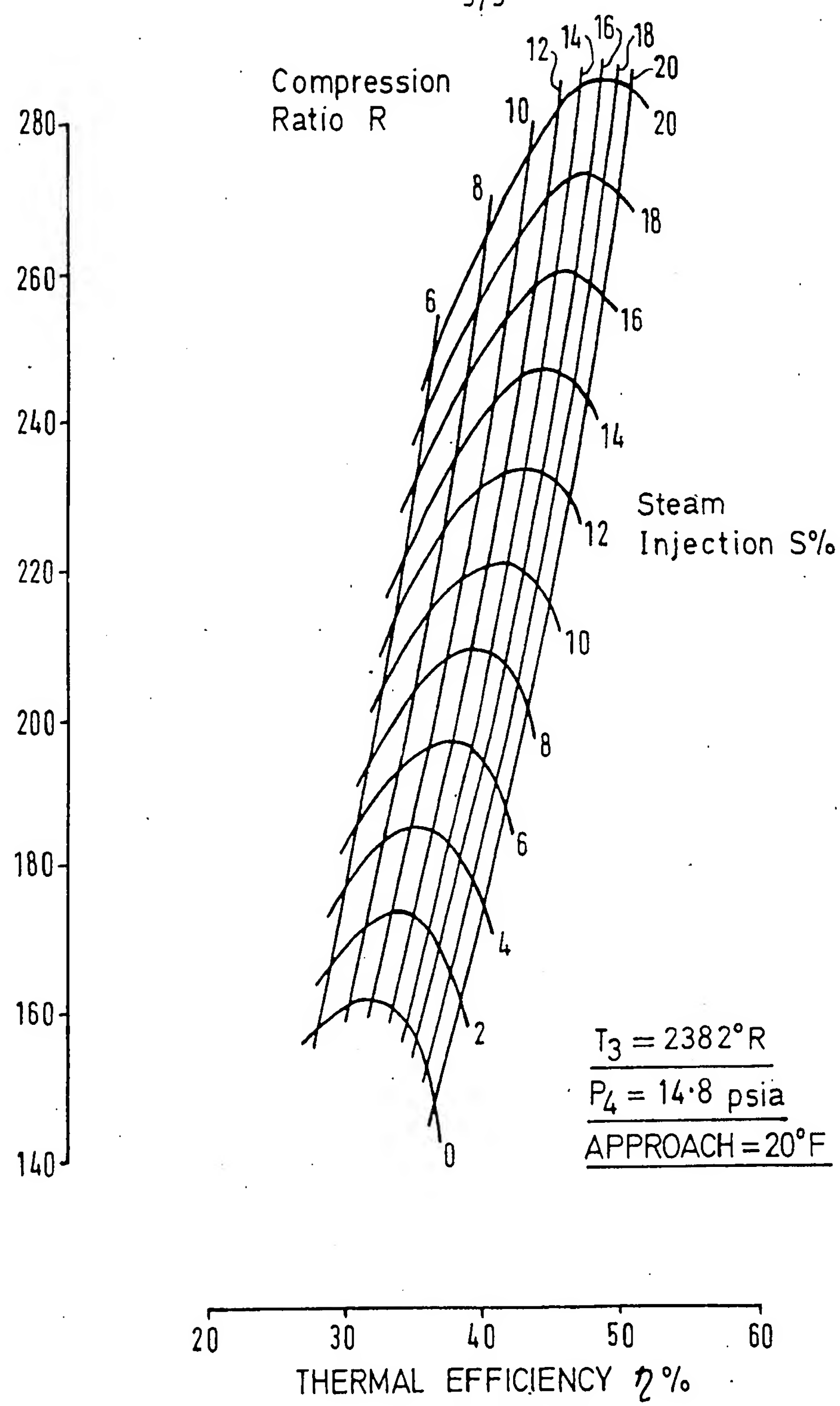
Compression
Ratio R
12
14
16
18
20
10
8
6
20
18
16
14
12
10
8
6
4
2
0
Steam
Injection S%
280
260
240
220
200
180
160
140
$T_3 = 2382°R$
$P_4 = 14{\cdot}8$ psia
APPROACH = 20°F
20
30
40
50
60
THERMAL EFFICIENCY η%

FIG.5.

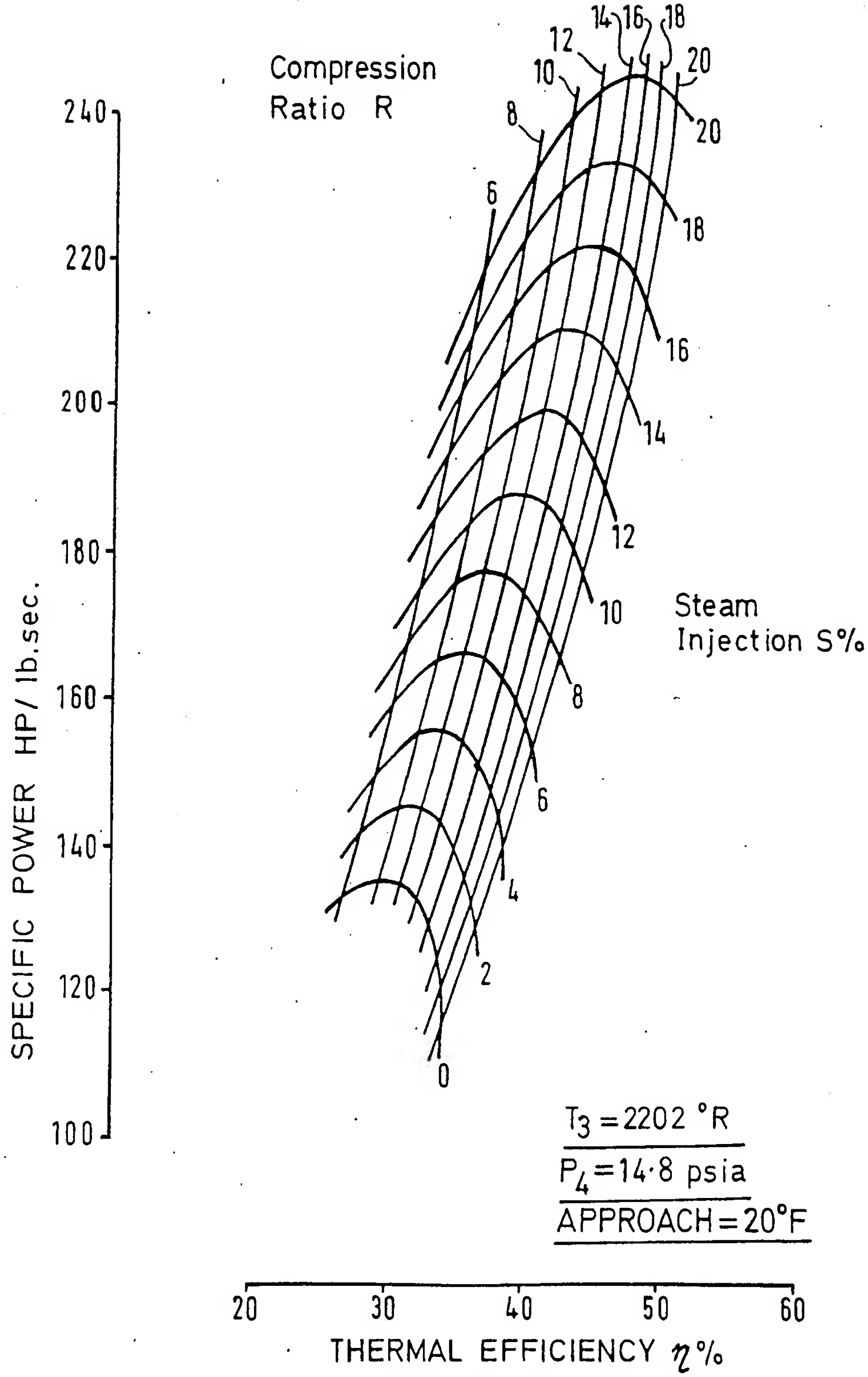
Compression
Ratio R
6
8
10
12
14
16
18
20
240
220
200
180
160
140
120
100
SPECIFIC POWER HP/ lb.sec.
20
18
16
14
12
10
8
6
4
2
0
Steam
Injection S%
T3 =2202 °R
P4=14·8 psia
APPROACH=20°F
20
30
40
50
60
THERMAL EFFICIENCY η%

FIG.6.

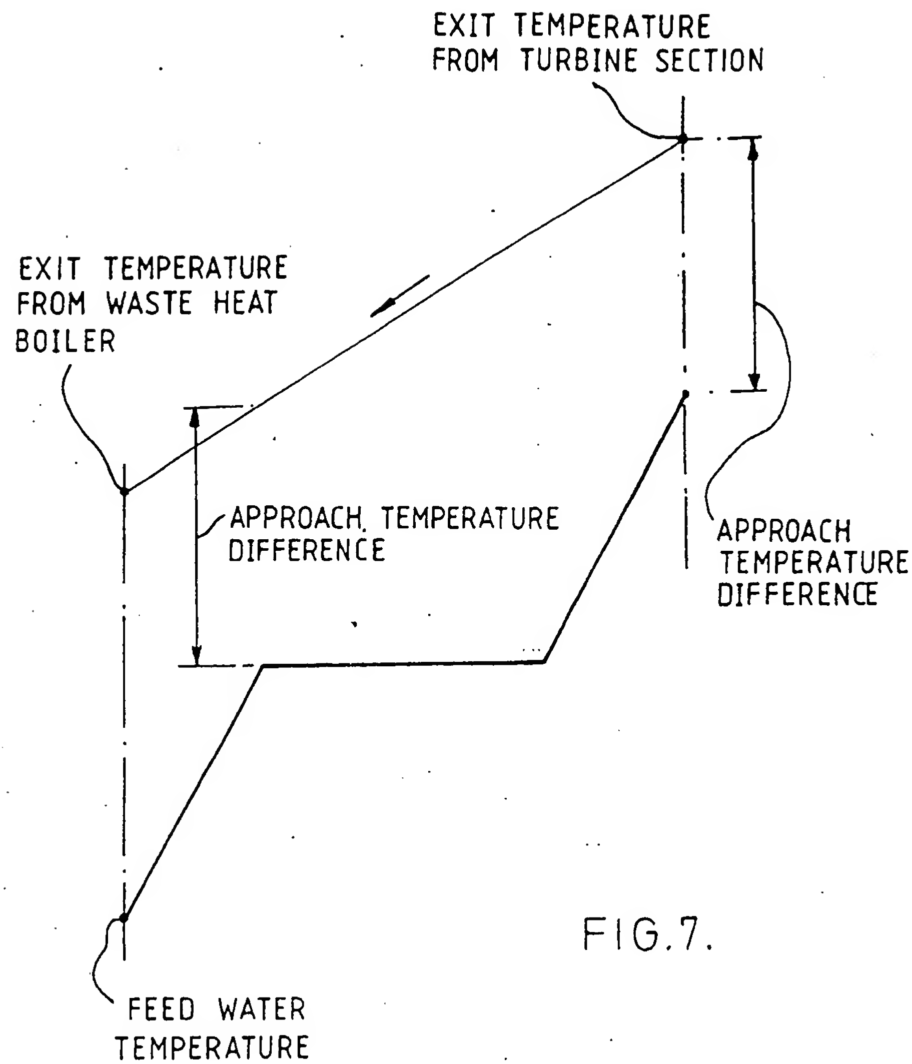
EXIT TEMPERATURE
FROM TURBINE SECTION
EXIT TEMPERATURE
FROM WASTE HEAT
BOILER
APPROACH. TEMPERATURE
DIFFERENCE
APPROACH
TEMPERATURE
DIFFERENCE
FEED WATER
TEMPERATURE

FIG.7.

SPECIFICATION

A gas turbine binary cycle

This invention is concerned with improvements in or relating to gas-turbine cycles for industrial power systems.

In the simple open gas turbine cycle, the processes of compression, combustion and expansion do not occur in a single component as they do in a reciprocating engine. They occur in components which are separate in the sense that they can be designed, tested and developed individually, and these components can be linked together in a variety of ways to form a gas turbine cycle.

The possible number of components is not limited to those already mentioned, other compressors and turbines can be added with intercoolers between the compressors, and reheat combustion chambers between the turbines. Heat exchangers may be included to recover some of the energy from the turbine exhaust gas by preheating the air before entering the combustion chamber.

The simple open cycle has a low efficiency due to high specific air flow and high exhaust temperature. The high air flow is required to provide tempering of the hot gases from stoichiometric temperatures to levels tolerable to turbine parts, and the high exhaust temperature is brought about by the low value of the expansion coefficient of the turbine. The various combinations of components are generally selected and arranged to improve the power output and thermal efficiency of the cycle. The way in which the components are linked together not only affects the maximum overall thermal efficiency, but also the variation of efficiency with power output and of the output torque with rotational speed.

For a gas turbine cycle using ambient air as the elastic working fluid, the way of obtaining increased efficiency is to:

a) increase values of the turbine inlet temperature, T_3, or

b) introduce means to recover some of the energy from the turbine exhaust gases.

Clearly T_3 is governed by the temperature limits of the highly stressed turbine blades. The maximum allowable value depends upon the creep strength of the materials used in the construction of the turbine and the working life required. Currently development is taking place in a variety of ways to be able to increase the working life by cooling blades, coating blades, using composite materials etc.

Development towards recovering waste heat has led to complicated arrangements which, although using well proved items of equipment, have detracted from the elegant simplicity of the classical simple open cycle, and in terms of size they have dwarfed the basic gas turbine, and costs per kW installed have increased.

The work output from a gas turbine depends upon the value of the exhaust temperature at the completion of expansion, which in turn depends upon the value of the specific heat ratio γ. Greater temperature drop is obtained from fluids with lower values of γ and it is apparent that the specific performance can be greatly improved by expanding a mixture of air and steam. When the injection steam is generated by the use of heat from the turbine exhaust gases then the cycle becomes thermally more efficient.

Figs. 5 & 6 illustrate the simultaneous increase in both thermal efficiency and specific power with variations of compression ratios and of steam injection ratios for fixed values of T_3 and of back pressure through the waste-heat boiler.

Inferior grade and relatively inexpensive petroleum fuel is widely used for combustion processes. Such fuels include petroleum liquid fractions which boil above about 800°F (about 425°C) and contain a substantial amount of ash-forming components. They include compounds of sodium, calcium, nickel, iron and from 2 to 400 parts of vanadium per million parts of fuel and also have sulphur concentrations in the range from about 0.5 to 5% by weight. When these fuels are burnt in gas turbine engines, they are excessively corrosive and have a high capacity for ash formation. It is known that, if fuel of this type is burnt with near stiochiometric quantities of air, i.e. 0 to 3% excess air, the corrosion and fouling characteristics are substantially decreased. In addition, when large quantities of air are used i.e. 5 to 500% above stoichiometric quantities, then ash is produced together with combustion products and they form deposits which corrode the hot parts of the engine downstream from the combustion chamber.

Stoichiometric combustion results in temperatures of about 3600°F (about 1980°C), which cannot be tolerated by turbine parts, and cooling down to about 1500°F (about 815°C) must be employed. Conventionally this is achieved by adding large amounts of excess air which require to be compressed and which produce the harmful ash and combustion products.

It is commonly known in conventional gas-turbine cycles for a compressor to accept air at atmospheric pressure and compress it to a pressure in the neighbourhood of 10 atmospheres (about 10 bars). The fuel providing the heat to operate the turbine cycle is burnt in the compressed air, thereby raising its temperature. The hot compressed gases are then expanded

through a turbine mechanically connected to the compressor and providing the driving power for it. In a typical cycle, the power capacity of the turbine is in excess of the power required by the compressor, and the surplus power is available for external use. In a second typical cycle the power requirements of the compressor and of the external load are each provided by matched turbines which are not connected to each other mechanically.

In both cases the gases as finally exhausted at atmospheric pressure are still at a fairly high temperature, and the heat in them is therefore available for the generation of power. It is an established practise to recover this heat by directing the exhaust gases to a heat exchanger in which they transfer this heat to the compressed air leaving the compressor and before the fuel is injected into this stream. This arrangement improves the thermal efficiency of the whole cycle but the maximum air temperature is unchanged and the temperature of hot gases entering the turbine is also unchanged. Furthermore the design of the heat exchanger is complicated by the high pressure of the compressed air stream, and the pressure drop generated by it in this stream is a further thermodynamic disadvantage.

It is further known to inject water into the gas stream in order to lower the maximum temperature reached in the cycle, and it is possible to reduce the work absorbed by the compressor by the injection of water. This injection of water has the disadvantages that solid particles in the water have an adverse effect upon the turbine parts onto which the gas stream impinges, and also creates surge conditions in the compressor eventually leading to stall.

Inherent in the simplicity of the open cycle and due to the metallurgical limitations imposed for practical operation are large specific air flow and exhaust temperature. The high air flow is required to provide cooling of the hot gases from stoichiometric temperatures to levels tolerable to turbine parts.

Stoichiometric temperatures for typical fuels are in the range of 3400°F to 4000°F (about 1870 to 2200°C) whilst turbine materials are limited to a gas temperature in the range 1400°F to 1800°F (about 760 to 980°C). Air flows many times the stoichiometric amount are required to lower stoichiometric temperatures to turbine temperatures. Flow values greater than 3 times the stoichiometric amount are common.

It is therefore an object of this invention to overcome some of the disadvantages of the above systems, and simultaneously improve both specific performance and of thermal efficiencies from typical values obtained from conventional gas-turbine cycles.

Thus there is provided a gas-turbine cycle in which turbine exhaust is used to generate steam which is then injected into a unique combustion chamber.

According to a further aspect of this invention there is provided a gas-turbine set incorporating a water-heat boiler and steam injector whereby steam can be generated at a pressure sufficient to permit its injection into the combustion chamber.

According to a further aspect of this invention there is provided an engine whose exhaust emissions have very low environmental pollution levels.

The proposed steam injection cycle is centred around the fact that combustion must be carried out with from at least 3% greater than stoichiometric air quantities, and that tempering of stoichiometric temperatures is carried out with steam to lower temperatures where the ash forming compounds will not oxidise even when more air is added to complete the tempering process.

Where steam is added in this fashion for cooling purposes and to provide a reducing atmosphere, it adds to the mass of working fluid without requiring the compression work characteristic of air cooling. This results in a cycle with increased power and improved thermal efficiency. The steam for injection is generated at a pressure slightly higher than the pressure inside the combustion chamber. This allows for pressure drop in the pipework and fittings. The generation of steam by the use of heat from the exhaust gases reduces the temperature of the normal dry exhaust gas and greatly increases the thermal efficiency of the cycle. However, in view of the presence of sulphur compounds within the fuel oil, the temperature reduction of the exhaust gas must be limited to avoid condensation. The gas within the preheater section should be kept above its acid dewpoint which is the highest temperature of surface on which an acid film will condense.

A limiting value of 300°F (about 150°C) has been used in the calculations, but it may be that the practical limit is below 300°F (150°C) due to the exhaust gases containing low concentrations of combustion products owing to the large amount of air used.

The exhaust gass, containing a mixture of combustion products, air and steam are expelled into the atmosphere at atmospheric pressure and about 300°F; thus, the injection of steam represents a steady consumption of water which is lost to the atmosphere.

With the extremely low steam pressures encountered in this cycle (150 psig—10.5 kg/cm²—for an engine with a compression ratio of 10:1) the required quality of feed water treatment is low. The amount of water vapour present in the exhaust plume will be approximately half the value of the water vapour leaving the cooling towers of modern power stations,

Reference will now be made to the accompanying drawings, of which

Figure 1 is a schematic cross section of a conventional gas-turbine combustion chamber employing downstream fuel injection, and

Figure 2 is a schematic cross section of a conventional gas-turbine combustion chamber employing upstream fuel injection, and

Figure 3 is a schematic cross section of a combustion chamber in accordance with the present invention.

A conventional gas-turbine combustion chamber as shown in Fig. 1 and Fig. 2 normally comprises three zones. In zone A fuel is mixed with air and combustion occurs with temperatures in excess of about 3000°F (about 1600°C). Excess air is passed over the inner wall which contains the combustion zone, thereby cooling it and reducing metal temperatures, and at the same time raising the temperature of the excess air. This hot air is then introduced into zone B and ensures that combustion is complete. The remaining air is introduced into zone C and cools the gases from zone B down to turbine temperatures for use in the gas-turbine cycle. Considerable power is absorbed by the compressor in compressing and delivering this cooling air.

During combustion of fuels with air in conventional gas-turbine combustion chambers, the following environmental pollutants are produced due to the large excess of air with which the combustion is associated.

NO_x: A small part of the nitrogen present in the air or in the fuel itself reacts with oxygen to form nitric acid in the flame gases. This nitric acid reacts further in the flame or when the combustion products leave zone A to form nitrous oxide (N_2O) and to a limited extent nitrogen tetroxide (N_2O_4). The mixture of these oxides of nitrogen is termed "NO_x".

SO_x: All petroleum products contain organo-sulphur compounds which are present as sulphides, disulphides or cyclic compounds. Their nature and concentration are dependant upon the origin of the crude oil, but the highest concentrations are found in residual fuels and typically in the U.K. the sulphur content ranges between 0.1% in kerosene to about 3% weight in heavy fuel oils. These sulphur compounds are rapidly converted to SO_x in the flame zone. The mixture of SO_2 and SO_3 is termed "SO_x".

CO: At flame temperatures the combustion products are disassociated, consequently, the products at the flame temperature contain small quantities of carbon monoxide, hydrogen etc. Since the carbon monoxide is formed rapidly in the reaction zone but only slowly consumed, the concentration of carbon monoxide present in the reaction zone is above the equilibrium value.

ASH: The most common constituents in fuel from which ash is formed are sodium, vanadium, calcium, magnesium, iron, nickel and silica, the first two being the most important. The melting points of sodium-vanadium compounds are:

α phase	V_2O_5	1244°F (673°C)
β phase	Na_2O, V_2O_4, $5V_2O_5$	1219°F (659°C)
γ phase	$5Na_2O$, V_2O_4, $11V_2O_5$	1071°F (577°C)
δ phase	Na_2O, V_2O_5	1166°F (630°C)

As the primary zone temperature is far in excess of these values, these sodium-vanadium compounds are rapidly formed.

It is therefore an object of this invention to overcome the disadvantages of burning fuels in conventional gas-turbine combustion chambers, and by controlling the atmosphere within the combustion chamber by limiting the amount of primary air in zone A of Fig. 3 to be not significantly less than 3% above the stoichiometric amount and by completely surrounding this near adiabatic flame by a continuous curtain of steam so as to prevent the formation of the pollutants previously mentioned. In this manner the atmosphere outside the flame but contained within zone A of Fig. 3 is completely free from air, thus adequately preventing any compounds of nitrogen, oxygen, sulphur, carbon, being formed by their absence. The steam used is at a much lower temperature than that of the combustion products and the amount of steam used is regulated to ensure that the temperature of the mixture of steam and combustion products is much less than the minimum temperature required to produce the sodium-vanadium compounds thereby preventing their formation. The remainder of the air is passed over the inner cylinder which contains zone A and zone C of Fig. 3, and in so doing becomes heated by the cooling effect it has on the inner cylinder wall, and is then introduced into zone D. The amount of this secondary air is regulated to ensure that the temperature of the mixture leaving the combustion chamber is at the correct temperature for use in the turbine section.

Referring to Fig. 3 exhaust generated steam is introduced into zone C such that intimate contact takes place at the adiabatic flame boundaries. Simultaneously the internal cylindrical walls are protected from the effects of the high flame temperature. The additional air required to reduce the gas temperature is introduced into zone D where cooling of the mixture of steam and combustion products from zone A takes place.

According to one aspect of the present invention there is provided a gas-turbine combustion

chamber comprising an air inlet, a steam inlet, a fuel inlet all at the entrance end of the combustion chamber. A certain amount of premixing of fuel with air, dependant upon the fuel characteristics, may with advantage be employed. The combustion chamber is of the concentric double wall type characterised in that:

(1) There is a flame shroud coaxial with the fuel and primary air supply, whose shape is shown in Fig. 3 as being frustro-conical, but any surface of revolution which generates a concave surface is adequate. Its larger end is the outlet for the combustion products and the surface area in contact with the combustion products is concave.

(2) The steam inlet is situated adjacent to the fuel inlet end of the combustion chamber and is arranged to discharge steam into zone C which is bounded in part by the convex surface of the flame shroud. This cools the flame shroud and flame tube.

(3) The flame tube is the inner of two concentric cylinders which together form part of the combustion chamber. Its internal surface is swept by a continuous curtain of steam, and this prevents any part of its surface from direct flame contact. Its external surface is swept by air and in so doing becomes heated by the cooling effect it has on the inner cylinder wall. This combination of air and steam cooling ensures that the inner cylinder has low metal temperatures together with a low longitudinal temperature gradient.

(4) The air inlet is arranged to provide both primary air for combustion and secondary air for cooling, and as shown in Fig. 3 the outlet from the compressor is connected to the annular passage between the two concentric cylinders. More than one connection may be required and although it is convenient to show it in Fig. 3 as radial and normal to the path which the products of combustion are directed, the air inlet may be arranged tangentially or have its axis coinciding with or parallel to the direction of the combustion products. Whichever method is employed, however, is dictated by the layout of the gas-turbine.

(5) The fuel supply into the combustion chamber, as shown in Fig. 3, is by a single burner. More than one burner may be required dependant upon the quahtity and type of fuel to be burnt. It is convenient to show only one burner in Fig. 3 as this simplifies the drawing and description thereof.

The unique combustion chamber and operation procedure will be readily understood by referring to the accompanying drawing illustrating one embodiment of the same. Refering specifically to the drawing Fig. 4, ambient air is drawn into the compressor 10 by means of line 1 where it is compressed and in accordance with the present invention, is introduced as primary air for combustion into the combustion chamber 20 by means of line 2. The amount of air supplied must not be significantly less than 3% greater than the stoichiometric quantity and at a pressure within the range 3 to 50 atmospheres (about 3 to 51 bars) such as about 10 atmospheres (about 10 bars). The combustion chamber 20 comprises an outer cylindrical shell 60 and an inner cylindrical shell 70, between which is an annular area 21. Within the inner shell 70 and connected to it is a flame shroud 80. The primary air is introduced into primary zone A between casing 70 and 80 of combustion chamber 20. Fuel is also introduced into primary zone A of casing 80 by conventional means within the casing 70 and 80. As mentioned, the amount of air introduced into zone A is always greater than the stoichiometric quantity.

In the present invention, steam is introduced into zone C which is bounded by the flame shroud 80 and the upper end of the cylindrical casing 70, which is conveniently shown in Fig. 4 as being a flat plate. This end closure may be ellipsoidal or hemi-spherical dependant upon the diameter of the outer cylinder 60 and the pressure of the air delivered by line 2. This steam is produced in a manner described hereinafter. This steam is introduced into zone C by means of line 4. This steam flows through the space 22 and serves to cool the outer or convex surface of the flame shroud 80 before being directed by the open end 23 of the space 22 to flow over the internal surface of the cylindrical shell 70 in a uniform and continuous manner. The combustion which occurs within zone A of casing 70 is highly efficient, and temperatures close to adiabatic values occur. The temperature of the casing 70 is kept to very low values by the passing of quantities of compressed air through the annular space 21 and the passing of steam through the open end 23 of the space 22. This additional amount of air is also delivered by means of line 2, and the compressor is sized to be able to deliver the amount of air for both combustion purposes and for cooling purposes. The steam issuing from the open end 23 is caused to mix with the combustion gases and this mixing causes the combined temperature to be in proportion with the amounts of the combustion gases and of the steam and their individual temperatures.

The hot mixture of combustion gases and steam flow into zone B wherein they are mixed before passing into zone D where they mixed with the air flowing through the annular space 21 which passes through slots, holes or louvres 5 which provide the path from area 21 into zone D. This mixing causes the combined fluids' temperature to be in proportion with individual amounts and temperatures. This added air serves as a quench for the mixture of combustion gases and steam, and the quantity is that required to produce a final temperature acceptable to

air delivered by compressor 10 is in excess of that required for the combustion carried out in zone A. This combustion is substantially complete.

The complete mixture of combustion gases, steam and air is delivered by line 6 into the turbine which functions in a conventional manner. The turbine 30 may be linked by suitable means 7 in order to drive the compressor 10. The load machine 40 is driven by suitable means 8 from turbine 30 as illustrated in Fig. 4. The arrangement shown in Fig. 4 is for a single spool turbine in which most of the pressure energy is expanded to do work, the exit pressure being higher than atmospheric by the amount of pressure drop required by the waste heat boiler. Twin spool turbines are commonly employed where the first spool is matched to the requirements of the compressor and the second spool matched to the requirements of the load machine and are deemed to be included although not shown in Fig. 4.

The turbine gases are exhausted from turbine 30 by means of line 9 and passed into the waste heat boiler 50 wherein they are employed to raise steam at a pressure sufficiently greater than that prevailing within zone A to enable it to overcome the resistance of line 4 and be passed into space 22. Sufficient water is introduced into the waste heat boiler 50 by means of line 10 while the spent exhaust gases are passed to atmosphere by means of line 11.

The spent turbine gases are exhausted from turbine 30 by means of line 9 and passed into the waste heat boiler and superheater 50 wherein they are employed to generate either saturated steam or superheated steam at a pressure sufficiently greater than that prevailing within zone A to enable it to overcome the resistance of line 4 and be passed into space 22. Sufficient water is introduced into the waste heat boiler 50 by means of line 10 while the spent exhaust gases are passed to atmosphere by means of line 11.

The advantage of the present invention is the increase in the efficiency of the cycle by injecting exhaust generated steam and simultaneously increasing the work output by increasing the mass flow without increasing the compressor. These advantages are clearly demonstrated by refering to Fig. 5 and Fig. 6. It is clearly seen for a classical open cycle i.e. the steam injection cycle operated with 0% steam injection, the profound effect that the work requirement of the compressor has on the overall performance. That is to say as the compression ratio is increased both power output and cycle efficiency reduce and further gains are only obtained by increasing the value of the maximum temperature i.e. the highest temperature at which expansion begins within the turbine section. With steam injection it is clearly seen how both efficiency and work output increase for fixed values of Tmax and of Compression Ratio, and it is clearly demonstrated how the previous dominance of the compressor requirements are overcome.

The waste heat boiler approach temperatures effect the cycle efficiency. Larger values reduce the cycle efficiency. Refer to Fig. 7.

As pointed out heretofore, the amount of air mixed with the fuel in primary zone A is always not significantly less than 3% greater than the stoichometric quantity.

The amount of steam introduced into the secondary zone B may vary and the amount selected is that which achieves the balance between the approach temperature and the exit temperature from the waste heat boiler. Depending upon the chemical composition of the fuel selected a lower limit is made for the exit temperature such that acid dew point conditions do not arise. This temperature is in the range 250°F–325°F (about 120–165°C) for fuels with a sulphur content of approximately 0.15% weight, and may be considerably less if this proportion is reduced. As pointed out heretofore the closer the steam temperature approaches the turbine exit temperature the greater is the value of the efficiency. Conversely larger waste heat boilers are required to achieve closer approach temperature, and an economic appraisal is necessary to select the optimum value of the approach temperature.

Values of steam quantities are plotted in Fig. 5 and Fig. 6.

The additional amount of air introduced into zone D is that necessary to reduce the temperature to that value at which expansion begins within the turbine section. This value is termed Tmax or T_3.

It should be noted that the exhaust gases contain large proportions of unburnt oxygen and no attempt is made to combust it by auxiliary firing. Thus the basic concept of the present invention is to burn typical distillate fuels, but not limited exclusively to their use, with small amounts of air which are always not significantly less than 3% greater than the stoichiometric quantity. The products from this combustion are too hot to be used directly in a turbine and must be cooled, and the combustion chamber must also be cooled. Exhaust generated steam is introduced into the combustion chamber wherein it cools the outer or convex surface of the flame shroud and thence flows over the internal surface of the cylindrical portion in a uniform and continuous manner thereby reducing the temperature, before mixing with the products of combustion. Such cooling may be described as boundary layer cooling. This produces a substantial cooling effect and ensures that further oxidation is prevented. This mixture is cooled to lower temperature by the addition of further air which has just been used to cool the outer surface of the cylindrical portion of the combustion chamber.

The curves in Figs. 5–6 show the variations of efficiency and of specific power for a wide

range of compression ratios and amounts of steam injection for a variety of maximum temperatures.

It is appreciated that adequate water treatment will be necessary to operate the waste heat boiler to prevent the possibility of the carry over of impurities into the turbine section where they might deposit. Such water treatment is conventionally prescribed.

It is a feature of this invention that the gas turbine can be started in the dry mode, and if required can be operated continuously in the dry mode, because it will require 15–30 minutes from lighting the burner before steam can be raised and thus injected into the combustion chamber.

CLAIMS

1. A method of operating a gas turbine cycle in apparatus including a zone in a combustion chamber into which fuel and air are fed for combustion, characterized in that steam is also fed into said zone of said combustion chamber so as to surround the resulting flame and to exclude air from the part of said zone about the flame.

2. A method according to Claim 1, in which the amount of air fed into said zone is at least 3% greater than the stoichiometric amount required for complete combustion of the fuel fed into said zone.

3. A method according to Claim 1 or Claim 2, in which said fuel is burnt substantially adiabatically.

4. A method according to any of Claims 1 to 3, in which said steam is generated by means of the waste heat of exhaust gases from said turbine.

5. A gas turbine having a combustion chamber which includes a zone where fuel combustion takes place to produce a flame, characterized in that there is also provided a steam generator and means for feeding steam from said generator to said zone so as to surround said flame.

6. A gas turbine according to Claim 5, in which means are provided to employ the waste heat of exhaust gases from said turbine to operate said steam generator.

7. A gas turbine according to Claim 5, substantially as hereinbefore described with reference to and as illustrated in Fig. 3 or Fig. 4 of the accompanying drawings.

8. A method according to Claim 1, substantially as hereinbefore described with reference to Fig. 3 or Fig. 4 of the accompanying drawings.

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